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79A24507
NASA TM-75757

STORAGE PEAK GAS-TURBINE POWER UNIT

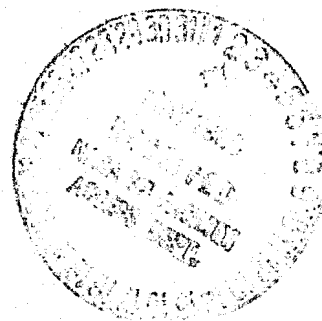
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Translation of "Akkumuliruyushchaya pikovaya gazoturbinnaya ustanovka",
Periodica Polytechnica, Mechanical Engineering, Vol. 22, No. 2 (1978)
pp. 95-113

(NASA-TM-75757) STORAGE PEAK GAS-TURBINE
POWER UNIT (National Aeronautics and Space
Administration) 25 p HC A02/MF A01 CSCL 10B

N80-21755

Unclas
G3/37 47670



NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
WASHINGTON, D. C. 20546 JANUARY 1980

STORAGE PEAK GAS-TURBINE POWER UNIT

by

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A storage gas-turbine power plant using a two-cylinder compressor with intermediate cooling is studied. On the basis of measured characteristics of a 25 Mw compressor computer calculations of the parameters of the loading process of a constant capacity storage unit (05.3 million cu m) were carried out. The required compressor power as a function of time with and without final cooling was computed. Parameters of maximum loading and discharging of the storage unit were calculated, and it was found that for the complete loading of a fully unloaded storage unit, a capacity of 1-1.5 million cubic meters is required, depending on the final cooling.

Introduction

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The storage peak gas-turbine unit is a promising aggregate of peak power in energy systems in light of the fact that the compressors are separated from the turbine, and they are driven by surplus and inexpensive night-time electricity, and not scarce and expensive hydrocarbons. Such aggregates can successfully compete with the hydro-storage plants to cover the peak power of the energy systems.

1. Compressor Group

Our studies were directed towards the peak gtu (gas-turbine unit) of the storage type consisting, according to the plan in fig. 1, of a low pressure turbine with power 180 mw, GT-100-750 unit of the IMZ [Leningrad Metal Plant] and gtu compressor with power 25 Mw. In the first variant a one-cylinder compressor is provided for without intermediate cooling (type G-9 of the Gërlitts Machine Construction Plant), and in the second variant--two-cylinder compressor with intermediate cooling (GT-25-750 of IMZ).

The first compressor with rated load has an air consumption of $G=154.3$ kg/s and degree of compression $\pi=13$.

* Numbers in margin indicate pagination in original foreign text.

Since the maximum pressure before the turbine in the beginning of unloading cannot exceed the permissible maximum pressure of 7.8 bar, with regard for the pressure loss the degree of compression of the compressor must not be greater than 9.34 if one has in mind storage at a constant pressure. With such a partial load the efficiency of the compressor is reduced from 0.849 to 0.781, the air consumption according to the compressor characteristics rises from 154.3 kg/s to 157.75 kg/s, the power will be 52.34 Mw, and the temperature behind the compressor will be 344.7°C.

These parameters are unfavorable from that viewpoint that due to the limited degree of turbine expansion the degree of compressor compression and its efficiency are reduced, and in addition during storage the high final temperature is lost.

In order to exclude these shortcomings further solutions were sought during which attention was drawn to the LMZ compressor of the G-25-750 type unit with power 25 mw, since its characteristics, thanks to the two-cylinder design with intermediate cooling are more favorable. This compressor with revolutions of 3000 1/min. for rated load according to the measurements of [4] has an air consumption of 197.4 kg/s, degree of compression of the low pressure compressor $\pi_1=3.23$, adiabatic efficiency 0.851, coefficient of coolant utilization 0.897, degree of compression of high pressure compressor $\pi_2=3.44$ and its efficiency is 0.847. /96

Since the total degree of compression during storage of constant pressure is somewhat high for the turbine, therefore the degree of compression needs to be reduced according to the measured characteristics respectively by $P_1=3.07$ and $\pi_2=3.16$. In this respect the efficiency of the compressor compartments is reduced by $\eta_1=0.8203$ and $\eta_2=0.8205$, the air consumption rises to 200.65 kg/s, the temperature behind the compressor due to the intermediate cooling is lower, a total of 180.3°C and power of the compressor 55.6 mw.

Of the two examined compressors we select the latter for the following

$$\bar{n} = \frac{\frac{n}{\sqrt{T_1}}}{\frac{n_u}{\sqrt{T_{1u}}}}$$

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yield the degree of compression of the compressor.

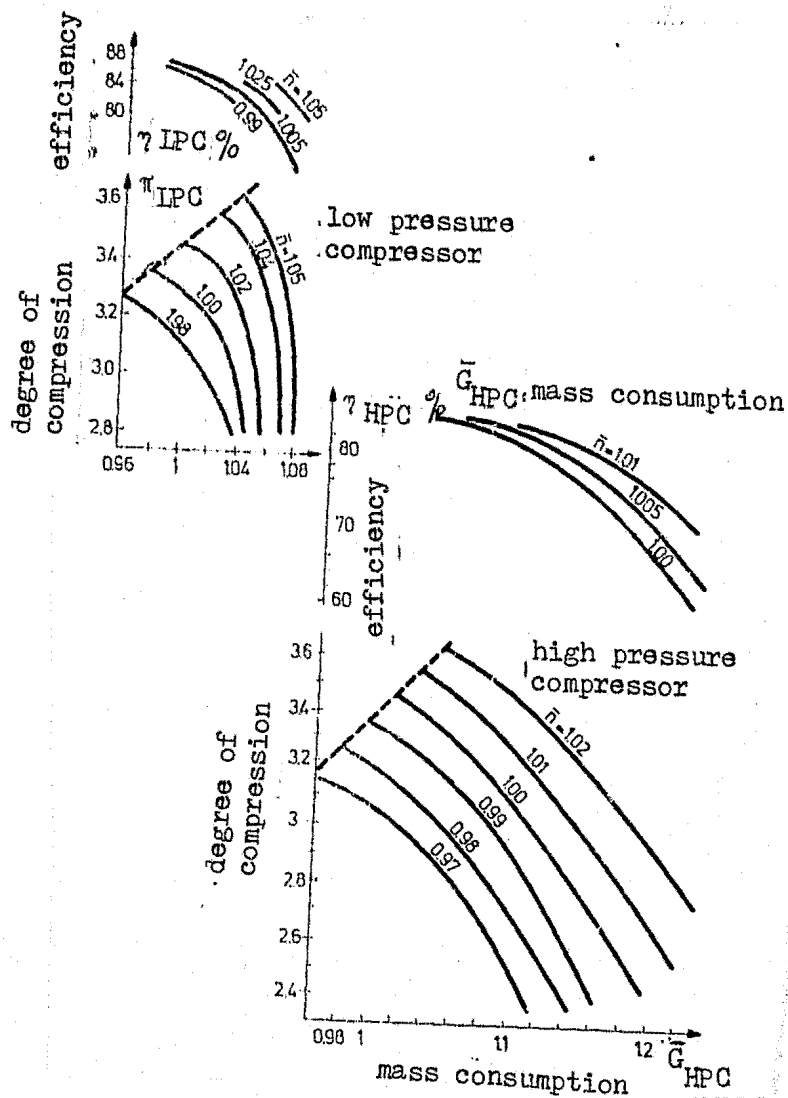


Figure 2. Measured Characteristics of Compressor

The nominal parameters of the low pressure compressor (IPC) are according to the designations in fig. 1.

$$G_H = 197.4 \text{ kg/s}, T_{1H} = 291^\circ\text{K}, p_{1H} = 0.996 \text{ bar}, n_H = 3000 \text{ l/min.}$$

The nominal parameters of the high pressure compressor (HPC) are the following: $G_H = 197.4 \text{ kg/s}$, $T_{2H} = 308.8^\circ\text{K}$, $p_{2H} = 3.085 \text{ bar}$.

To compute the process of loading it is necessary to have the total degree of compression of the compressor group $\pi = p_2/p_1 = \pi_1 \cdot \pi_2$ and the adiabatic efficiency of the compressors in the function from the mass consumption.

The IPC and HPC characteristics are not directly summed, since they can operate only with intermediate cooling which needs to be considered.

We compute the intermediate cooling on the basis of data from measurements by the constant coefficient of pressure loss $\sigma_{\text{cool.}} = 0.963 = \text{const}$ and the temperature behind the compressor is computed and the conditions for constancy of the utilization coefficient of Boshchnakovich (ψ). The compressor of the GT-25-750-1 type unit has a surface of intermediate coolant of 3400 m^2 , and according to the data of measurements in function of the consumption of cooling water the utilization coefficient is almost not altered, thus, with consumption of cooling water 1000 T/h $\psi = 0.88$, with consumption of 2500 T/h $\psi = 0.91$ since the water equivalent of the cooling water is almost an order greater than the water equivalent of air. Therefore in our computations we assumed the condition of constancy of ψ .

According to this condition with respect to the known temperatures at the inlet of air and cooling water one can determine the air temperature behind the intermediate cooler. The rated temperature of the cooling water $T_H = 295.5^\circ\text{K}$ and the rated coefficient of utilization of the heat exchanger $\psi_H = 0.8968$.

The adiabatic efficiency of the entire compressor group is computed according to the following relationship (from the designations in fig. 1):

$$\eta = \frac{\eta_1(T'_1 - T_1) + \eta_2(T_2 - T'_2)}{(T'_1 - T_1) + (T_2 - T'_2)}$$

Compilation of the total characteristics of the compressors was carried out according to the following technique.

Throttling occurs on an air filter therefore at the compressor inlet we have $T_1 = 288^\circ\text{K}$, $p_1 = 0.97$ bar with coefficient of pressure loss of the filter $\sigma_f = 0.97$.

The IPC characteristics:
mass air consumption

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$$G_1 = G_{1m} \cdot \sqrt{\frac{T_{1m}}{T_1} \frac{p_1}{p_{1m}}} \quad G = 193.25 \text{ G [kg/s]}$$

the relative number of revolutions

$$\bar{n}_1 = \sqrt{\frac{T_{1m}}{T_1}} = 1.005$$

From these values for the IPC characteristics one can find the degree of compression and efficiency.

The specific work of compression of the IPC

$$L_1 = C_p T_1 \left(\pi_1^{\frac{\kappa-1}{\kappa}} - 1 \right) \frac{1}{\eta_1} \quad [\text{kJ/kg}]$$

Temperature behind IPC

$$T'_1 = T_1 + \frac{L_1}{C_p} \quad [^\circ\text{K}]$$

Pressure behind IPC

$$p'_1 = \pi_1 \cdot p_1 \quad [\text{bar}]$$

Parameters for intermediate cooler, since

$$\eta = \frac{T'_1 - T_2}{T'_1 - T''_1} = \text{const.}$$

Temperature at outlet

$$T'_2 = T'_1 - \psi(T'_1 - T_n) = T'_1 - 0,8968(T'_1 - 295,5) \text{ [}^\circ\text{K]}$$

Pressure at outlet

$$p'_2 = p'_1 \sigma_{\text{cool}} = 0,963 \cdot p'_1 = \pi_1 \cdot 0,97 \cdot 0,963 = 0,9411 \pi_1 \text{ [bar]}$$

After this according to the known p'_2 and T'_2 based on the condition of constancy of mass consumption $G_1 = G_2$ one can compute the criteria of similarity from the HPC characteristic (fig. 2), then from them one can define π_2 and η_2 .

Based on this the specific work of compression of the HPC

$$L_2 = C_p T'_2 \left(\pi_2^{\frac{\kappa-1}{\kappa}} - 1 \right) \frac{1}{\eta_2} \text{ [kJ/kg]}$$

The temperature at the outlet from the HPC

$$T_2 = T'_2 + \frac{L_2}{C_p} \text{ [}^\circ\text{K]}$$

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pressure for the HPC

$$p_2 = p'_2 \pi_2 \text{ [bar]}$$

The given mass consumption of air of the total characteristic $G = G_1$ since the condition of the inlet and the rated parameters are the same.

The total degree of compression

$$\pi = \frac{p_2}{p_1} \sigma_{\text{cool}} \cdot \pi_1 \cdot \pi_2$$

The total specific work of compression:

$$L = L_1 + L_2.$$

The total characteristics of the compressors determined thus are shown in figure 3.

In the process of charging of the storage the working point of the compressor will be moved along the curve $\bar{n}=\text{const.}$ between certain lowest and upper limits. The upper limit is defined by the boundary of the compressor pumpage. The lower limit of the working zone is defined by the interruption in the stream in the last stage of the compressor with large negative angles of incidence or jump in the thickening in the last stage. This lower boundary is usually not shown on the characteristics of the compressors since during joint operation of the compressor and the turbine the lower boundary is defined by the idling of the gas-turbine unit and the curve of the idling is always above the lower boundary curve. In the storage gtu the lower boundary of the stable operation is the actual boundary.

The upper boundary of stable operation with the aforementioned calculated revolutions $\bar{n}=1.005$ according to the total characteristics of the compressors compiled from the individual measured characteristics passes with degree of compression $\pi_{\text{max}}=10.83$, while the lower boundary according to the evaluation of the measurements of idling with $\pi_{\text{min}}=4.9$.

The total characteristic of the compressor (fig. 3) in machine computations was considered by the following approximate quadratic equations:

$$G = -8,9325 \cdot 10^{-4} \pi^2 + 8,7539 \cdot 10^3 \pi + 1,029$$

$$T_2 = -14750 G^2 + 29752 G - 14538,5$$

$$P = -28400 G^2 + 57580 G - 28904,18$$

Although the GT-25-750-1 compressor can operate only with intermediate /101 cooling this circumstance is favorable during storage in a constant volume since due to the intermediate cooling the temperature for the compressor is reduced, and this is advantageous because in the given volume of storage the mass of stored air is increased with lower energy outlays of compression. One should bear in mind that the hot compressed air during storage with time is cooled, and this heat loss in the combustion chambers needs to be additionally reported therefore it is expedient to use intermediate cooling which according to the calculation results reduces by 8-12% the specific work of compression the temperature for the compressor by 22-25%.

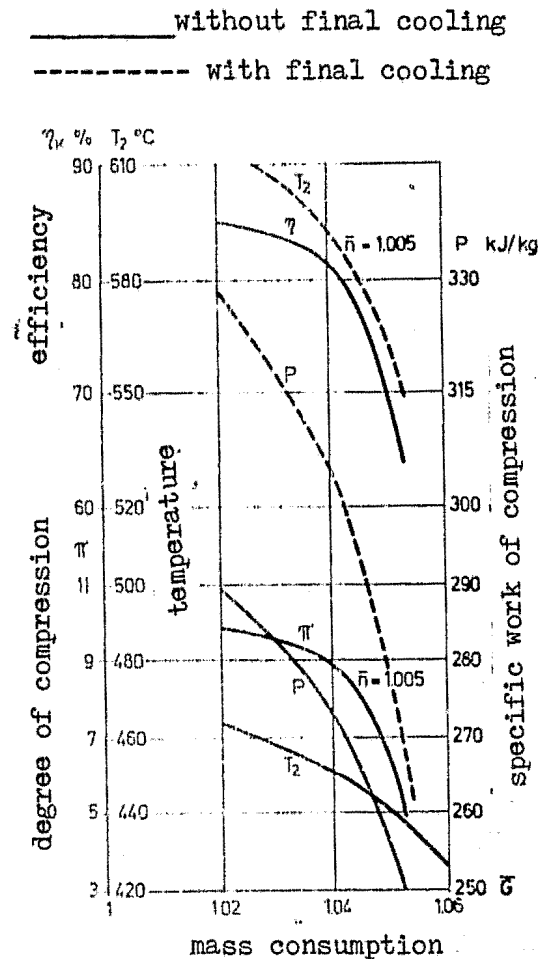


Figure 3. Total Calculated Characteristics of Compressors

2. Loading of Storage

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In the process of loading, the compressor from the external electric drive loads the constant capacity storage unit. Here the state of the inlet is not altered, while the pressure for the compressor constant rises, therefore the working point on the characteristic curve of the compressor is altered all the time.

This change in the working point is approximately considered in the following model of machine computation.

In the computation model we consider the working point unchanged for the period of time Δt during which the forced air increases the pressure in the storage unit, while this determines the new working point. The upper

and lower limit of computation is determined by the limit degrees of compression π_{\max} and π_{\min} . With pressure of the storage unit p_x the degree of compression of the compressor $\pi = p_x/p_1$. Based on this from the total characteristic (fig. 3) we obtain the amounts G, η, L and T_2 . The mass air consumption is determined by the relationship $G = 193.25 \cdot \bar{G} \text{ (kg/s)}$. We define the new working point at the end of the period Δt based on the hypotheses of complete mixing of the air charge of the storage unit and the supplied air in the period of time Δt . The equation of mixing

$$m_x \cdot T_x + G \cdot \Delta t = (m_x + G \Delta t) T$$

where m_x -- mass of air in storage.

Equation of state of the ideal gas:

$$p_x \cdot V = (m_x + G \Delta t) R T_x$$

From here we will express the pressure in the storage unit

$$P_x = \frac{m_x G \Delta t}{V} R \frac{m_x T_x + G \cdot \Delta t T_2}{m_x + G \Delta t} = \frac{R}{V} (m_x T_x + G \Delta t \cdot T_2) \quad (\text{bar})$$

The power consumed by the compressor:

$$N = L \cdot G.$$

At the end of the time period Δt the new degree of compression is

$$\pi = \frac{P_x}{P_1}$$

The computation begins with π_{\min} and continues to π_{\max} , and during the computation we obtain the change in parameters and power of the compressor depending on the time according to figure 4.

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The energy consumption of the compressor is obtained by multiplying the power of the compressor by the time interval and the subsequent addition of the results.

In principle, storage can be of constant pressure or constant volume.

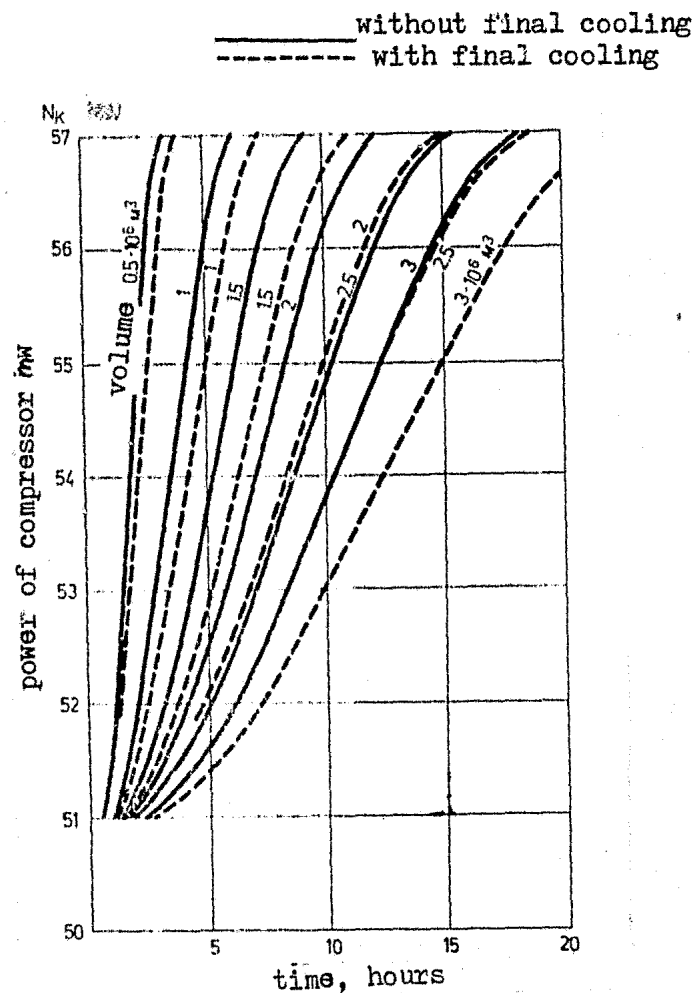


Figure 4. Power of Compression during Loading

In the case of storage with constant pressure the storage unit is an enormous connected vessel which requires considerable quantity of buffer water close to the storage unit. Such a storage is more complicated and more expensive due to the buffer system and is linked to the available water resources.

The necessary volume of the storage unit for the examined gas turbine has an order of a million cubic meters. It is expedient to employ a natural water storage unit, abandoned shaft or natural gas well. In the first case due to the wide wells the aerodynamic resistance will be low, /104

therefore in the calculations we assume that during loading and unloading of the storage unit the pressure loss occurs only due to the drop in air temperature during storage in a constant volume. Further it was assumed that in the shaft there is no place for leakage of compressed air.

The volume of the storage unit was considered to be a variable parameter and the computations were made for volumes $(0.5-3) \cdot 10^6 \text{ m}^3$ with spacings of $0.5 \cdot 10^6$ cubic meters.

Study of the storage in wells of natural gas is more complicated since storage occurs in porous rock where with diffusion movement of air a pressure loss develops. With such storage the residual natural gas is displaced by the air and such storage can affect the extraction of oil and gas of the nearest fields.

If the storage unit is the volume of a shaft and we employ subsequent cooling for the compressor, then in this cooler on the part of the air the process of throttling occurs according to the law $p_3 = p_2 \cdot \sigma_{\text{cool}}$, where $\sigma_{\text{cool}} = 0.97$ coefficient of pressure loss of the cooler. The total ratio is provided by the correlation of p_2 and p_1 . If the storage pressure is p_x then

$$p_2 = \frac{p_x}{\sigma_{\text{cool}}} \quad \text{and} \quad \eta = \frac{p_x}{p_1 \cdot \sigma_{\text{cool}}}$$

Since $p_1 \cdot \sigma_{\text{cool}} = \text{const}$, further the calculation is made in the same way as in the case of the absence of subsequent coolant. It is necessary to also consider that the temperature T with subsequent cooling is replaced in the previous formulas by the place T_2 . Subsequent cooling should be carried out until the temperature of the storage unit is reached, since only thus can one avoid a drop in pressure as a consequence of air cooling in the storage unit. Subsequent cooling in the computations is considered by the condition $T_3 = \text{const}$.

For the machine computation of the process of loading the time interval was selected at $\Delta t = 900 \text{ s}$, temperature of the storage unit 353°K and it

was assumed that with subsequent cooling the temperature T_3 will be this same amount.

The result of the machine computation of the loading process with subsequent cooling and without it is presented in figures 4-8. Figure 4 shows the required power of the compressor with respect to the time with the parameter of the storage unit volume. In light of the fact that loading of the storage unit is economical with inexpensive night-time electricity (generated by coal power plants), and the period of loading can be estimated as 6-8 h the given compressor requires a storage unit volume of no less than $(1-1.5) \cdot 10^6 \text{ m}^3$ since the smaller volume is loaded by the compressor in a shorter time than the time of the nocturnal dip. The storage unit with greater volume is loaded by the given compressor during the nocturnal dip only in the case of incomplete unloading, which is expedient due to the higher mean power of unloading. /105

In a comparison of the processes with subsequent cooling and without it it is found that with subsequent cooling the loading time rises for any volume since the storage temperature, according to our conditions, remains constant.

With subsequent cooling there is a rise in the mass of the air loading. This growth with the minimum volume of the storage unit is 8.42-9.13%. The required power of the compressor in both cases is roughly 53.6 mw, therefore due to the great time of loading the energy consumption for compression during subsequent cooling is above by 15.1-23.4%. Additional energy consumption for storage with subsequent cooling of 1 kg of additional air mass with minimum volume of the storage unit is 0.086-0.087 kw-h.

At the end of the process of loading the storage unit is closed until the beginning of the peak period. During this time the differences in pressure and temperature of the compressed air are balanced and the air temperature approaches the temperature of the storage unit wall.

In our computed model instantaneous current mixing in the storage unit is provided for, and in the case of the absence of subsequent cooling the

the temperature of the forced air is higher than the temperature of the walls, which results in cooling of the air in the storage unit. This cooling depends on the storage time, on the geometry and material of the storage unit, etc. For a specific storage unit this cooling can be determined by measurement depending on the time. In an extreme case during prolonged storage the air is cooled up to the wall temperature. In light of the lack of detailed data we consider this most unfavorable case.

In the case of subsequent cooling we assume the equality of the temperature of the air and walls of the storage unit which means that cooling is absent during storage and the maximum pressure $p_3 = p'_3 = 10.19$ bar.

In the absence of subsequent cooling the air with constant volume is cooled to the temperature of the walls, whereby the pressure will be

$$p'_3 = p_3 \cdot \frac{T'_3}{T_3}$$

At the end of loading the air temperature practically does not depend on the volume of the storage unit since the deviation from the mean amount does not exceed 0.3%. The mean air parameters: $T_3 = 399.96^\circ\text{K}$, $p_3 = 10.5$ bar, $T'_3 = 353^\circ\text{K}$. In light of this the pressure at the end of storage $p'_3 = 10 \cdot 353 / 399.96 = 9.27$ bar still is higher than the permissible pressure before the turbine even with regard for pressure loss; this means that in the initial stage of unloading it is also necessary to throttle the cooled air. /106

3. Turbine Aggregate

As a turbine of the peak storage gtu we will examine the low pressure turbines (IPT) GT-100-750-2 produced by LMZ with rated power 179 mw. Such turbines are in operation at the Inota power plant and in the future can be constructed. The five-stage single-housing IPT has the following rated parameters:

$$G_{TH} = 467.67 \text{ kg/s}, p_{6H} = 7.8 \text{ bar}, p_{7H} = 1.05 \text{ bar}.$$

The IPT is easily disconnected from the gas turbine unit since it has a separate shaft with coupling on the side of the IPC and bearings so that

one can use a synchronous generator directly for the drive. The selection of the given turbine is also governed by its considerable unit power.

The results of measurements [2] of the working process of the turbine are classified with a narrow interval of the stable working point of complete loading, and in the plan of the storage unit the turbine will operate due to the nature of the unloading in very broad limits. In light of the absence of experimental data the dependence of the pressures on the mass consumption was determined according to the Stodola formula:

$$\left(\frac{G_T}{G_{Tn}} \right)^2 = \frac{p_6^2 - p_7^2}{p_{6n}^2 - p_{7n}^2} \frac{T_{6n}}{T_6}$$

From here we will express the mass gas consumption:

$$G_T = G_{Tn} \sqrt{\frac{p_6^2 - p_7^2}{p_{6n}^2 - p_{7n}^2}} \sqrt{\frac{T_{6n}}{T_6}}$$

where the index "H" refers to the known rated pattern. With regard for the rated parameters we obtain

$$G_T = \sqrt{3661,27 p_6^2 - 4036,55} \quad [\text{kg/s}]$$

We assume that the temperature before the turbine for the entire period of unloading is regulated at $T_6 = T_{6H} = 1023^\circ\text{K}$ since this is a thermally expedient and technically feasible solution. In such a case the temperature correction in the Stodola equation $T_{6H}/T_6 = 1$.

Such temperature regulation in the initial aggregate is unfeasible because a drop in the power is possible only with a drop in the temperature before the turbine since the IPC with constant numbers of revolutions has a roughly constant air consumption, and this consumption is affected by the pattern conditions of the HPC on the free shaft only insignificantly.

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Regulation with a constant temperature before the turbine is a comparatively simple task in the storage unit since here the operating interrelationships of the compressors are lacking. Such regulation is advantageous since here the thermal efficiency of the cycle during unloading is

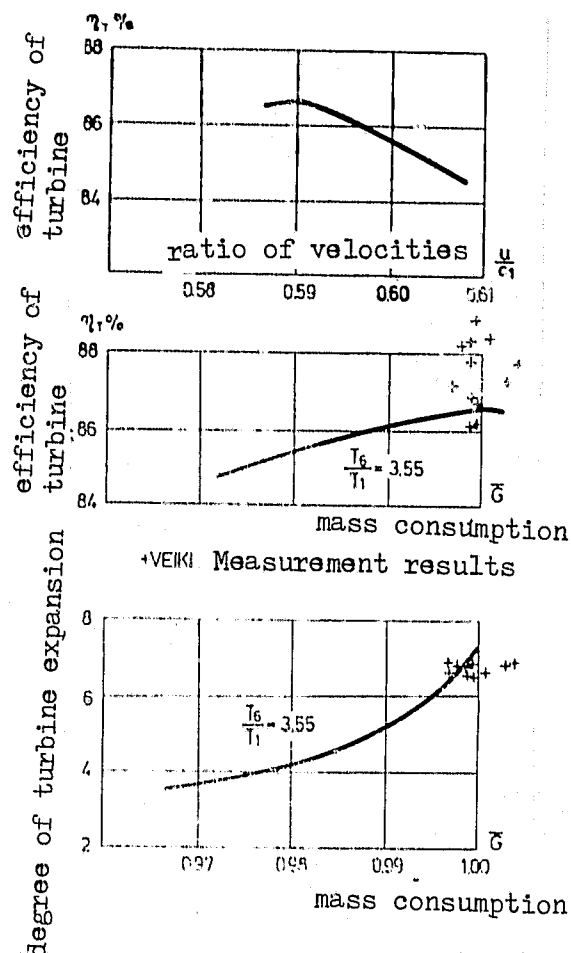


Figure 5. Characteristics of Turbine

reduced to a lesser degree than with a variable temperature.

With such regulation the coefficient of air surplus in the combustion chamber will be constant since the temperature before the combustion chamber is also constant, equal to $T_5 = 353^\circ\text{K}$. The constant coefficient of air surplus $\alpha = 3.4$, the fuel consumption therefore is directly proportional to the air consumption.

Based on the adopted and computed data the characteristics of the turbine are presented as standard dimensionless criteria of similarity in fig. 5.

The adiabatic efficiency of the turbine in the function of the mass consumption G was computed according to the known diagram $\eta_T = f\left(\frac{u}{c_1}\right)$ since the peripheral velocity (u) of the turbine is known, while the velocity of the emergence from the jet grid c_1 is determined according to the known section of emergence and the unknown mass consumption.

Besides the calculated curves of consumption and the efficiency figure 4 showed the results of measurements from [2], which yielded scattered points around the calculated amounts. A certain deviation is explained by the discrepancy in the ratio of characteristics of temperatures at the points of measurements from the calculated values $T_6/T_1 = 3.55$.

In machine computation the conditions of consumptions were considered by the Stodola equation, while the turbine efficiency was considered by the quadratic equation of approximation:

$$\eta_T = -3,023645 \cdot 10^7 G_T^2 + 2,828137 \cdot 10^{-4} G_T + 0,799645$$

which at the calculated points yielded a deviation of not more than 0.6%.

4. Unloading of the Storage Unit

Unloading of the storage unit by means of consumption of compressed air occurs in time with a reduction in pressure in a nonstationary process, the working point of the turbine is constantly altered, but in the computation model in a short period of time Δt we consider it constant, like the process of loading. If the pressure in the storage unit is p_x then the degree of expansion of the turbine with regard for the coefficients of pressure losses of the air filter $\sigma_f = 0.97$ and the combustion chamber $\sigma_{c.c.} = 0.927$.

$$\delta = \frac{p_0}{p_1} = \frac{p_x \cdot \sigma_f \cdot \sigma_{cc}}{p_1}$$

Based on the degree of expansion of the turbine with respect to the characteristics of the turbine (fig. 4) with known parameters before the turbine p_6 and T_6 we determine the mass consumption of gas and the efficiency of the turbine. According to these data the specific operation of expansion and the power of the turbine is determined which after multiplication

by time intervals Δt and subsequent summing yields the produced energy (fig. 8).

At the end of the period Δt the pressure in the storage unit is determined based on that condition that in this period the mass of compressed air is reduced by the amount $G \cdot \Delta t$. We assume that the unloading is not too fast, and therefore during the unloading the temperature in the storage unit is not altered (isothermic unloading). If in the beginning of the period Δt the mass of compressed air is m_x then

$$m = m_x - G \cdot \Delta t \quad p_x = \frac{mRT}{V}$$

From here the entire computation occurs in the same way as was evident /109 in loading. The upper and lower limits of unloading are determined by the pattern conditions of the compressors, the upper limit is determined by the pumpage of the compressor, while the lower limit is determined by the minimum pressure of the re-triggering of the compressor. From the viewpoint of the turbine the maximum pressure with temperature before the turbine 750°C equals 7.8 bar, the minimum pressure is limited only by the condition of re-loading of the compressor, it is not limited by the turbine.

Since the compressor can create pressure that is greater than is permissible for the turbine therefore in unloading it is necessary to throttle to the maximum pressure before the turbine $p_{6\max} = 7.8$ bar. The throttling is carried out until the pressure in the storage unit equals

$$p_{x,op} = \frac{p_{6\max}}{\sigma_c \cdot \sigma_{cc}} = \frac{7.8}{0.97 \cdot 0.927} = 8.67 \text{ bar}$$

The time necessary for throttling with volume of $0.5-3 \cdot 10^6 \text{ m}^3$ according to figure 8 showing the pressure of unloading with respect to time during storage with final cooling is 0.5-3 hours which is a comparatively small percentage of the complete time of unloading (2-12 hours) especially without final cooling.

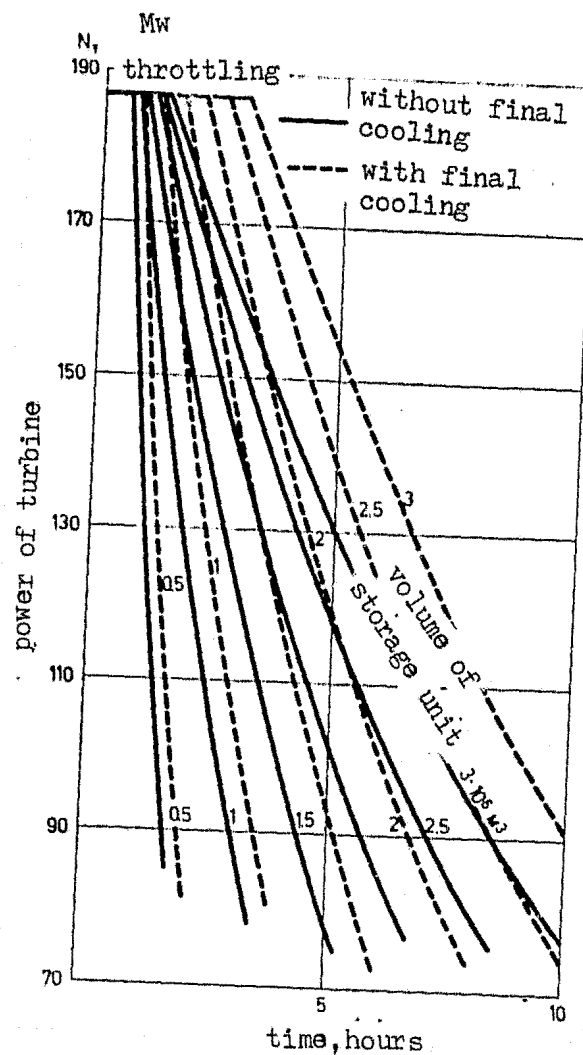


Figure 6. Power of Turbine during Unloading

We note that the computation of the process was made not quite to the pumpage boundary of the compressor since one needs to have a reserve of stability with respect to pumpage 5-7% according to the expression

$$\frac{\pi_p}{\pi} \frac{G}{G_n} - 1$$

where the index "n" refers to the pumpage pattern of the compressor.

The minimum volume of the storage unit $(1-15) \cdot 10^6 m^3$ from the viewpoint

of the complete use of loading during the nocturnal dip is also the minimum from the viewpoint of unloading since according to figure 8 we can determine that the complete time of unloading is 3.5-5.3 hours per day which is already sufficient for the goals of peak power plants. The storage unit with large volume either is unloaded completely further (with $3 \cdot 10^6 \text{ m}^3$ in 12 hours) or with shorter unloading is not completely unloaded, but only partially, and as a consequence of this the power with respect to time is reduced to a lower degree.

The results of the machine computation of the unloading process of the storage unit with final cooling and without it are shown in fig. 5-8, including figure 6 which shows the change in the power of the turbine, figure 7--the change in production of electricity, and figure 8--the change in air pressure depending on time. /110

By comparing the unloading of the storage unit with the two methods of loading it is found that the time of unloading of the storage tank during loading with final cooling is greater (which without final cooling and with the minimum volume of the reservoir is only 3.2-4.7 hours), therefore from the storage unit with final cooling we obtain more electricity, as a consequence of the fact that here the storage unit will contain a greater mass of compressed air in the beginning of unloading and operates in broader limits of pressure change. /111

The power of the turbine during unloading has the nature of a decrease; according to figure 6 with a completely loaded storage unit with power roughly 180 mw it is reduced to 80 mw in the unloaded state, whereupon in the beginning of unloading during throttling it is constant. During storage with final cooling this decrease in power is less as a consequence of the greater time of unloading.

Summary

We examined the variant of peak storage gas-turbine unit consisting of a IPT unit GT-100-175-2 produced by LMZ with maximum power 180 mw and gtu compressor type GT-25-750 of the LMZ included in a plan according to /112

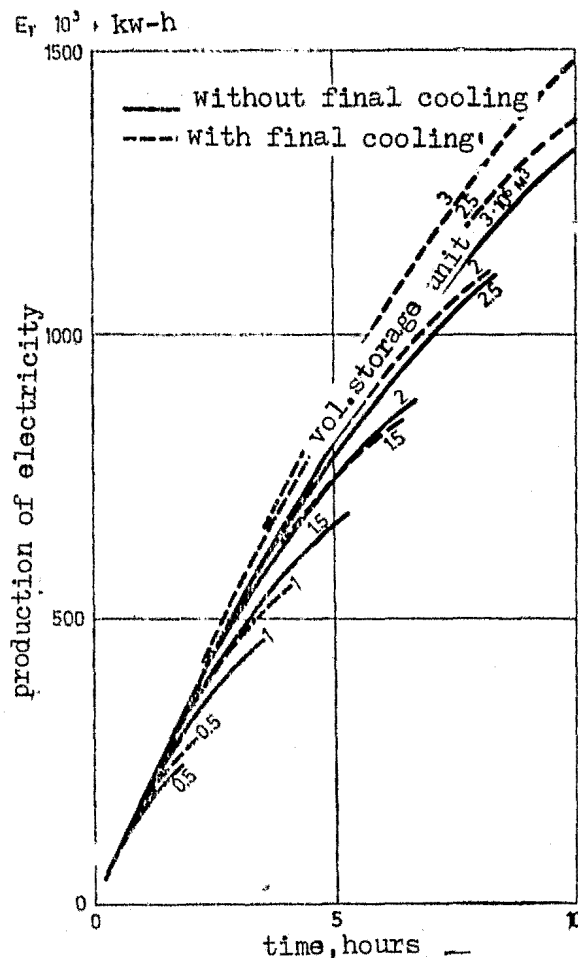


Figure 7. Production of Electricity during Unloading

figure 1. Based on the computations it was established that as a compressor of such a unit it was more expedient to use a two-cylinder compressor with intermediate cooling of type GT-25-750 of the IMZ with power 25 mw than a one-cylinder compressor without intermediate cooling of the gtu of type G-9 of the same power.

Based on the measured characteristics of the compressor GT-25-750 by the method of machine computation the parameters were determined for the process of loading the storage unit of constant volume in the interval

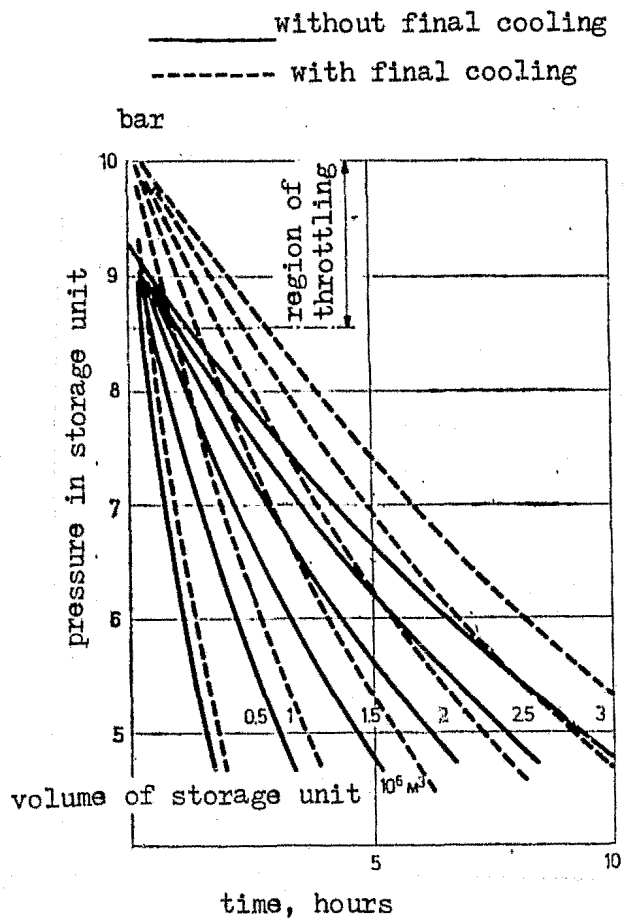


Figure 8. Pressure in Storage Unit during Unloading

$(0.5-3) \cdot 10^6 \text{ m}^3$ and the required power of the compressor with respect to time with the use of final cooling and without it.

Parameters were computed for the maximum loading and unloading of the storage unit based on the operating conditions of the compressor. As a result it was determined that for complete loading of a completely unloaded storage unit depending on the final cooling a volume of $(1-1.5) \cdot 10^6 \text{ m}^3$ is required. The processes of loading and unloading were determined for a storage unit of type mine wells whose computation as a result of wide sections is simpler than the computation of a porous storage unit of the type abandoned natural gas wells.

The method of machine computation was used to determine in the process of the proposed isothermic unloading the power of the turbine and the pressure in the storage unit with respect to time with the parameter of the storage unit volume in the case of loading with final cooling and without it.

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